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Solar cooling systems for climate change mitigation: A review

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ABSTRACT

The impact of global warming and an increase in the indoor cooling equipments using energy sources other than conventional energy have become very attractive because they can reduce consumption of fossil fuels as well as harmful emissions in to the environment. The solar energy is one of the readily available forms of renewable energy which can be used to operate the cooling equipments depending on the geographical location of the area where solar cooling system needs to be installed. The effectiveness of solar cooling also needs to be evaluated based on various performance indicating parameters. However, different types of solar collectors also need to be evaluated in order to find out their feasibility for cooling applications. Thus in this article review of different types of solar cooling technologies have been carried out. The study reveals that evacuated tube collectors are best option for solar cooling where as desiccant cooling helps in improving the indoor air quality. Also, thermal energy storage and ejector based solar cooling efficiently improves the performance besides energy saving.

Contents

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1. Introduction

The electricity is one of the readily used energy source for heating, ventilation and air-conditioning in both working and living environments. About one third of the total electricity is used for air-conditioning and refrigeration purposes [1]. Also, about 80% of the electricity so generated globally comes from the oxidation of fossil fuels causing many environmental problems [2]. The increased use of air conditioning and refrigeration operated by electricity has also promoted the use of fossil fuels for the production of electricity which in turn increases the global warming. Therefore, it is quite urgent to minimize the consumption of fossil fuels and to promote renewable energy sources like solar energy for refrigeration and air-conditioning. The concept of solar energy for cooling application was suggested 35 years ago as a solution to cooling problems in dry and desert areas where electricity was not readily available and the energy crisis in 70's has again attracted the attention of scientists and engineers to develop cooling machines which can work efficiently and economically with solar energy [3]. The various cooling techniques are confined with two major challenges which include the phasing out of Chloro–Fluoro Carbon based refrigerants and to curb the consumption of high grade electricity besides water as cooling medium [4]. It is also well appreciated that with gradual technological developments the solar energy turns out the best complimentary to conventional energy. The phasing out of R-12 (Dichlorodifluoromethane) refrigerant has led to the synthesis of the R-22 (Chlorodifluoromethane) as a substitute, but it is also associated with the problem of higher compression work and high operating pressure which can be overcome if it can operate at condenser temperature below the ambient [5,6] and therefore, solar-driven ammonia–water vapour absorption systems help in overcoming the problem. Different authors worked on the choice of working pairs for absorption systems driven by thermal energy sources and also various demonstration projects have been launched to gain more experience in the design and operation of solar refrigeration and air-conditioning [7–11]. It is also stressed that for the continual population and economic growth, wider use of solar energy in air-conditioning would secure the increasing energy demand. Among various technologies, absorption refrigeration has been most frequently adopted technology for solar cooling. It however, requires very low or no electrical input and also shows the high heat and mass transfer coefficient. Although a number of studies have been reported employing various absorption cycles [12–17] but some disadvantages of traditional absorption refrigeration systems, such as a low coefficient of performance (COP) and strict demand of heat supply, both in quality and quantity, have not been overcome yet. In particular, as the solar input constantly varies, indicating that solar powered refrigeration system will not be able to work consistently in the day which may lead to the deterioration of the performance of the system. Therefore certainly, there is a need of modifications in the absorption systems so that their performance can be improved effectively.

2. Different types of solar cooling systems

2.1. Absorption cooling systems

2.1.1. Test and simulation of a solar-powered absorption cooling machine

Alfred Erhard and Erich Hahne [18] carried out the simulation study of the solar powered absorption cooling machine suggesting the use of solar cooling in countries with a high solar energy supply. The workers also stressed on the fact that absorption cooling machines with solid absorbent are the best suited in hot and dry areas than wet absorption machines because such machines are easy to handle and have no moving parts, therefore do not require extra cooling equipment for high ambient temperature. However, as a drawback, these machines alternatively produce absorption (cooling) cycle during the night and desorption (heating) cycle during the day. Therefore, a cold storage like ice or some cold mass has to be provided for daytime cooling to power a cooling system. However, solar-powered cooling systems using a solid sorbent have frequently been discussed by many researchers [19–21]. An analysis of a solar-powered dry absorption cooling system with a collector integrated reactor was carried out [18]. The reactor consists of two steel pipes insulated with steel wool, filled with the absorbing medium, namely strontium chloride ($SrCl_2$). About 15% of graphite is also added to improve the thermal conductivity of the $SrCl_2$ and to prevent the salt from caking. The working of the cycle can be divided into four phases. Desorption (high-pressure phase) followed by an intermediate cooling phase and the absorption phase (low-pressure phase) followed by an intermediate heating phase. The reactor during the day is heated and acts as desorber which means that ammonia is driven out of the compound $SrCl_2 \cdot 8NH_3$ and stored in the degassing pipe. After sun set, the reactor works as an absorber and cool down until the absorption temperature is reached where the absorption takes place. The heat of absorption is transferred into ambient by means of pipes operating in a reverse way resulting in the temperature drop in the cooling compartment. A simulation study for a process is carried out based on a data base containing physical properties of all materials and working fluid pairs. A model program has also been developed for the cooling machine and working fluid in order to calculate the temperature and concentration in the desorber/absorber. The results obtained from the simulation study indicate about the system pressure and ammonia mass inside the ammonia reservoir. The results obtained also revealed that ammonia mass increases in the morning and reach maximum in the evening when heating process ends. There occurs a steep decrease of ammonia mass followed by an increase at the beginning of absorption process which probably originates from the pressure influence increasing up to 14 bars. During absorption process the pressure decreases and matches the pressure at the evaporating temperature. The temperature inside cooling compartment does not rise above 6 °C during the whole measurement period. However, during the desorption process the temperature inside the reservoir increases which is found to be slightly higher than ambient temperature because of the fact that the condenser temperature is above the ambient temperature. The Coefficient of performance obtained is quite less but with certain modifications in the system there could be an improvement in the performance.

2.1.2. A new absorption refrigeration cycle using solar energy

Chen and Hihara [22] proposed and evaluated a new refrigeration cycle driven by heat and electricity using $LiBr–H_2O$ as working fluid as shown in Fig. 1. The electricity operated compressor is augmented in the system so that a constant energy input to the system can be maintained. Many authors have performed many modifications like introduction of thermal storage, electricity

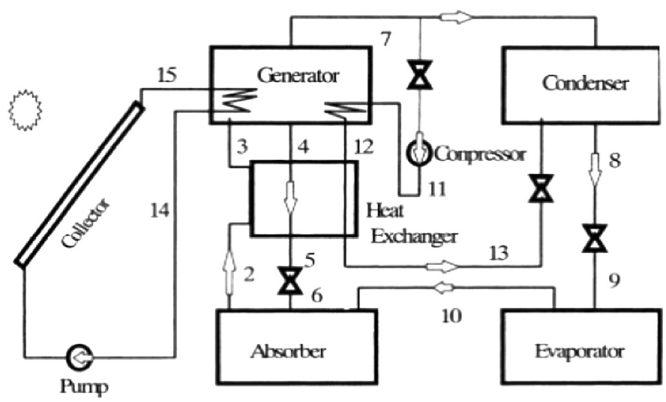

Fig. 1. Systematic scheme for the new refrigeration cycle [22].

driven vapour compression systems [23,98] but these are quite complex and also less economic. The authors therefore, added a compressor to the traditional vapour absorption cycle. This can also be done with the double effect system but the compressor needs to be provided at lower pressure generator side. The refrigerant vapours are divided in to two parts one is passing directly in to the condenser and other is passing through the compressor to raise the temperature which can be taken by the heat exchanger contributing more heat exchange in to the solution which can reduce the heat load of the generator. Such system is beneficial in two ways- (a) the heat cannot be removed directly in to the environment through condenser as in the conventional systems and, (b) reduces the temperature difference between the heat transferring fluid and environment. These benefits can lead to the reduction in irreversibility besides an increase in the performance of the system because of higher rate of utilization of the energy. The thermodynamic properties of the fluid were taken from the ASHRAE standards. The heat losses and losses due to the resistance of flow in the conducting pipes were neglected in the study. The various assumptions were considered in the analysis which include daily trend of solar intensity, hot water temperature, isentropic efficiency of the compressor (0.7), saturated nature of the vapours in the generator after getting heat in the generator, the saturated nature of the refrigerant and weak solution leaving the condenser and absorber at 40 °C. The different performance parameters were calculated based on simulation study using the assumptions discussed above. The results obtained indicate that theoretical minimum temperature obtained at evaporator is the defining parameter for refrigeration and only if the evaporating temperature is higher than minimum temperature, the positive refrigeration effect is produced. This minimum temperature depends on the concentration of the strong solution leaving the generator and the temperature of the solution in the absorber. The refrigeration capacity and COP are important parameters in the analysis as revealed form the study and the obtained results stated that COP calculated in the study is found to be quite higher than the conventional system because of higher rate of energy utilization in the new system. It also becomes evident from the obtained results that the condenser heat load is also important parameter as it clearly identifies the operational cost of the system and is found to be very small in comparison to traditional cycle especially when solar intensity (heat supplied to the collector) is small.

2.1.3. *Optimum hot water temperature for absorption solar cooling*

The Lithium Bromide–water solution is the most commonly used as the working fluid in absorption machines. Therefore, Lecuona et al. [24] carried a study to find optimum hot water temperature for absorption solar cooling. Gordon and Ng [25] attribute external irreversibilities to heat transfer in the heat exchangers. Many authors [26–28] carried out a review of various models that included solar collector in the analysis and asserted that there is an increase in chiller efficiency with an increase in generator temperature but also results in a decrease in solar collector efficiency showing that there exists an optimum generator temperature for maximum collector efficiency. The effect of the generator temperature is obtained for a variety of cycles, solar collector configurations and solution compositions in a solar cooling layout without a heat storage tank [29]. An algorithm based on characteristic equation model was considered that could yield optimum hot water temperature needed for the generator. The resulting algorithm is simple as it requires only measurement of the external temperatures to the chiller. The solar collector efficiency operating under steady state conditions is modelled with linear equations for obtaining the results. The analysis stated that the there exists a hot water temperature below which the machine is not capable of cooling but consumes heat because of internal losses. The study also revealed that to maintain maximum generator temperature corresponding to maximum solar coefficient of performance along with time requires control of the mass flow rate through the collector. This study also represents installations in which a bypass on the storage tank hot water inlet is used for control and optimization [30]. The model of the characteristic equation, based on the characteristic temperature, allows for development of an explicit equation for the optimum hot water temperature in solar cooling. It maximizes the instantaneous efficiency in solar cooling installations for single-effect LiBr–H_2O absorption machines. Losses in the solar collector field and corrections for operational differences from standard test conditions have been considered in the analysis. The model assumed temperature jump in the hot water which results from the temperature difference. Fortunately, the optimum hot water temperature is weakly dependent on the value of this parameter, making an iterative optimization approach possible over successive short time intervals, progressively correcting this parameter. This equation is an approximation that can be applied in two cases

a. When a satisfactory global fit of the characteristic equation constant parameters to real COP data of the absorption machine is sufficiently accurate, and
b. when these parameters must be considered as functions of heat rejection and evaporator external average temperatures Thus, it offers a broader applicability than conventional characteristic equation based optimization. This modified characteristic equation has been fitted to the COP data of a rotary-type chiller, yielding good results. The application of the developed equation requires several online temperature measurements, solar irradiance and wind speed. To reach the optimum hot water temperature, flow control in the primary circuit is required. The variation in flow rate has also been studied for hot water storage and non-storage configurations.

2.1.4. *First and second law analysis of a new power and refrigeration thermodynamic cycle using a solar heat source*

Hasan et al. [31] carried out the first and second law analysis based on simulation study to find out the optimized conditions of a power cycle and thermodynamic refrigeration cycle using solar energy. The analysis is carried out with the optimization program GRG2 which uses the generalized reduced gradient (GRG) algorithm to optimize the cycle [32]. The simulation is carried out for the investigation of the performance over a wide range of heat source temperature (330–470 K). It has also been revealed that for the sensible heat sources like solar thermal systems or waste heat

from the exhaust gases etc., Lorenz cycle is an ideal cycle giving the best performance [33]. Cengel and Boles [34] also stated that efficiency of the thermodynamic cycle depends on the fraction of input energy converted in to useful energy output and also compared the performance of the cycle with an ideal reversible cycle. The variables used in the simulation are boiler exit temperature, rectifier exit temperature, super heater exit temperature, system high pressure, heat source inlet temperature, heat source flow rate and effectiveness of recovering heat exchanger. The low pressure of 2 bars and ambient temperature of 290 K was kept fixed for simulation. The simulation program neglects the pressure drop, heat losses and also considers the pump as well as turbine as 100% efficient. It was also assumed that the turbine exit temperature had to be 280 K or lower with a vapour mass fraction (dryness fraction) of 90% for the refrigeration effect. The simulation is carried out for the heat source inlet temperature of 400 K and accordingly the various performance parameters were calculated. The simulation results showed that first and second law efficiencies lie in the range of 16.9 and 65.4% respectively. The results obtained from the study also revealed that ammonia vapours expands through turbine from 2.05–0.2 MPa (at pressure ratio of 10.2) to a temperature of 280 K which is lower than the assumed value of 290 K. The mass fraction of ammonia in the basic solution leaving the absorber is 0.437. A large fraction of the basic solution returns from the boiler to the absorber through Solution heat exchanger. The net amount of work obtained is 16.9% and the corresponding refrigeration produced is 1.26%. The highest irreversibility is found to be in absorber which is 44% where as in rectifier and Solution heat exchanger the irreversibility is found to be 16 and 24% respectively. The results obtained also stressed that the maximum second law efficiency can be obtained at heat source temperature of 420 K and above this efficiency drops gradually. The study revealed that the heat transfer rates in a cycle are the function of the heat source temperature and heat transfer in the rectifier increases with an increase in the heat source temperature, which eventually increases the irreversibility in the rectifier. The pressure ratio also increases in the heat source temperature and at the heat source temperature of 470 K the pressure ratio obtained is 27 which indicates that such ratio of pressure ratio requires more than one expansion device. The results from the study also point that irreversibility in the cycles is mainly due to the absorber, recovery heat exchanger and rectifier and such losses could be reduced by using different refrigerant mixture and also through better matching of heat source temperature in the boiler.

2.1.5. Modelling, simulation and optimization of a solar collector driven water heating and absorption cooling plant

Sorption systems, either by absorption or adsorption processes are utilized to produce cooling. Vargas et al. [35] carried the modelling, simulation and optimization of a solar collector driven water heating and absorption cooling plant as shown in Fig. 2. However, efforts in that direction have been made by author [36] showing an improvement in specific cooling power (SCP) besides achieving a coefficient of performance of 0.39 through the results obtained from numerical simulation and optimization. Different authors [27,37–39] have obtained typical values of COP of around 0.15 for the solar energy based heating and cooling absorption systems. The experimental study of the two stage adsorption systems was also carried out along with different cycles (e.g., continuous heat recovery, mass recovery, thermal wave, cascade multi-effect, hybrid heating and cooling) [40]. The work done by Wang [41] reported values for the COP close to 0.5. The simulation study of cooling system based on trough and evacuated flat plate collector combinations to investigate the reduction of gas-firing requirements in single and double-effect absorption lithium bromide chillers have also been carried out [42] and the findings showed the largest potential savings of 86% for the double-effect chiller and trough collector. Hasan and Goswami [43] presented an exergy analysis of a solar driven power and refrigeration cycle and found that increasing the heat source temperature increases first law efficiency along with larger exergy destruction in the system. The heat driven refrigeration systems have been proven feasible and practical, but low COP and large size are hurdles still need to be overcome. Thermodynamic optimization is one possible way of achieving better performance and size reduction in order to make these systems commercially competitive. The method of entropy generation and minimization has emerged during the last three decades as a distinct sub field in heat transfer [44,45]. However, Vargas et al. [35] proposed a model for solar collector and gas driven water heating and refrigeration plants, based on the fact to extract maximum exergy input from solar collector or fuel sources. The modelling is done for a system having three integrated subsystems- (I) solar collector (II) energy storage tank and hot water heat exchanger, and (III) regenerator heat exchanger and absorption refrigerator. The net heat transfer rate for the collector surface is calculated as suggested by Buzelin et al. [46] which were based on energy balance. In this type of model all variables were directly proportional to actual dimensionless model and any one of the variable used for simulation could easily represent the entire model. The optimization is carried out based on mass flow

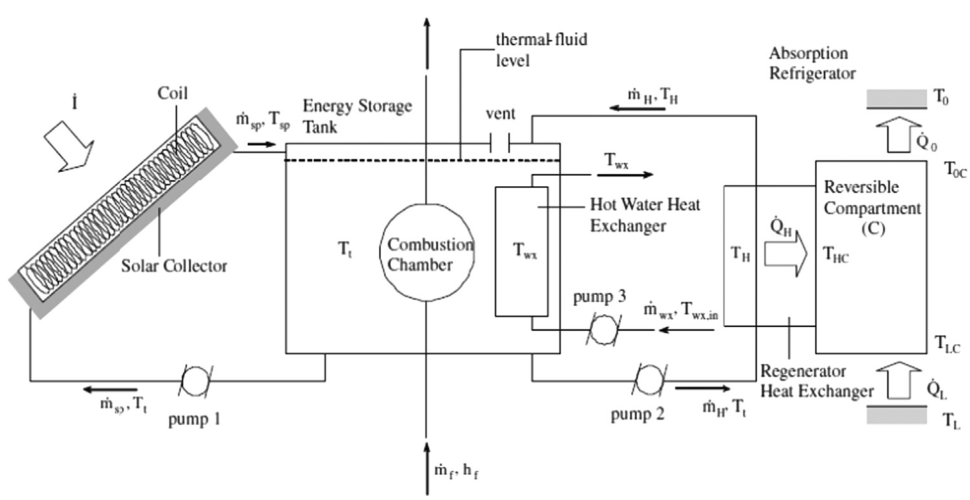

Fig. 2. Schematic diagram of the proposed solar and gas fired driven water heating and refrigeration plant [35].

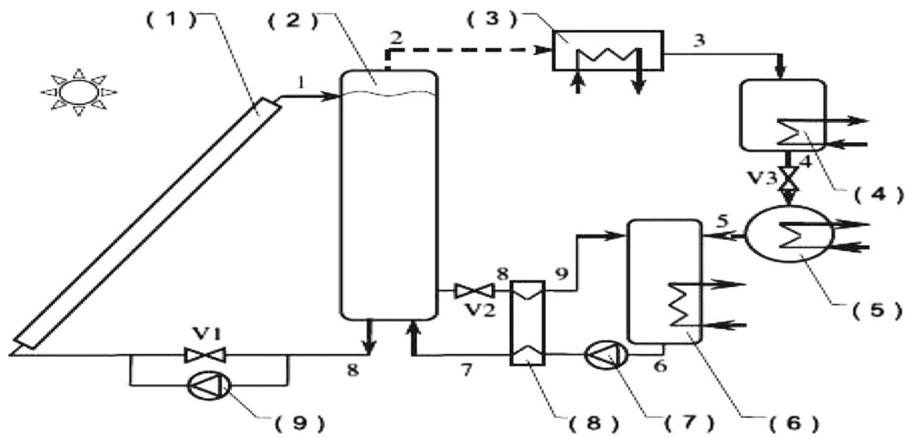

Fig. 3. Schematic diagram of SPAR system with VMETS technology using LiBr–H_2O as working fluid indicating different components as (1) Evacuated solar collector with a metallic absorber, (2) Solution storage tank, (3) Condenser cooled by cooling air or cooling water, (4) Water storage tank, (5) Evaporator (6) Absorber, (7) Solution pump, (8) Solution heat exchanger, (9) Auxiliary pump and V1, V2 and V3 are the controlling valves [48].

rate which states that when mass flow rate approaches zero, the heat transfer rate as well as exergy approaches to zero whereas when mass flow rate approaches to infinity, the cooling as well as heating rate is negligible because the temperature of working fluid passing through the heat exchanger is almost same as the inlet temperature.

The study was carried out for minimizing the pull-down and pull-uptimes to reach specified set points, i.e., a refrigerated space temperature, and a hot water temperature respectively. The numerical method calculates the transient behaviour of the system using equations which were integrated using an adaptive time step, 4th–5th order Runge–Kutta method [47]. It is also worthy explaining that when sun radiation dropped to zero, the system solely operates on the heat transfer rate of combustion chamber and when sun's radiation come back to half of maximum intensity the system again restores its working from the solar collector energy. The heat transfer rate in thermal fluid comes to be 50% of the net solar collector radiation transfer rate indicating more losses at higher collector temperature, which need to be accounted in the system design. The evaluation of the refrigerator considered in the study revealed that COP comes to be 0.5 which is a typical value in absorption refrigeration. The optimization with respect to the collector thermal fluid capacity rate is also calculated both for minimum pull-down and pull-up times and also for heat capacity rate in the regenerator (thermal fluid) as well as hot water heat exchanger (water) which is calculated as 0.239 & 0.5 respectively. It has also been observed that the optimal value of the thermal fluid in collector is the same for both cooling and heating because both hot water heat exchanger and the refrigerator are driven by the same heat source. The optimization was also done separately for cooling and heating for better interpretation of results. The dimensionless regenerator heat capacity rate was kept fixed (0.239) and the dimensionless water capacity rate was varied with regenerator heat capacity rate. The minimization of the pull-up time with respect to regenerator thermal fluid capacity rate for three values of dimensionless thermal fluid capacity rate of water in hot water heat exchanger results in the calculation of dimensionless heat capacity of the thermal fluid in the coil as well as regenerator which comes to be 1.73 and 0.239 respectively. It was also concluded that minimized pull-up time decreases monotonically as dimensionless thermal fluid capacity rate of water in hot water heat exchanger decreases. This behaviour is different from what was observed in the refrigerated space pull-down time. In the cold space, the thermal load is kept fixed in the optimization process therefore the pull downtime is minimized. Conversely, in the hot water heat exchanger, the load decreases as the water capacity rate decreases, therefore the less water the faster it is to heat it up. However, although the water stream is heated up faster, less exergy is available, i.e., the potential for use of the hot water reduces. The major conclusions drawn from the study indicate that there is a fundamental optimal heat capacity rate in the solar collector and gas fired driven water heating and refrigeration plant that determines the water heating rate as well as refrigeration produced. The optimization also revealed that maximum second law efficiencies are found to be sharp, stressing their practical contribution in the design to make such systems commercially more competitive. Also, the thermal collector fluid capacity rate is found to be of the highest importance when compared to dimensionless thermal fluid capacity rate of water in hot water heat exchanger and regenerator, an important finding for the purpose of system scalability.

2.1.6. An investigation of the solar powered absorption refrigeration system with advance energy storage technology

An increase in the use of indoor cooling equipment has promoted the application of renewable energy to power air-conditioning systems and has extensively reduced the global warming as well as harmful emissions. Xu et al. [48] carried out an investigation of the solar powered absorption refrigeration system with advanced energy storage technology as shown in Fig. 3. The analysis is carried out because the fluidity of the absorbent gives greater flexibility for a more compact and an efficient system [10]. Presently, the solar powered absorption refrigeration system is a very attractive application of solar energy though solar energy input varies constantly, indicating that solar powered refrigeration will not be able to work with consistency throughout the day. The field test results of a solar cooling plant along with the performance in terms of energy collected and the temperature of the solar collector was also presented by Lazzarin et al. [49]. They suggested two measures i.e., installing a direct fired absorption system or to install electricity driven vapour compression system which can keep the cooling capacity steady but such measures also enhances the operational cost. This problem can be overcome by constructing a thermal storage system [23,50] but the low energy density storage restricts its application in solar cooling. Thus an advanced energy storage technology called Variable Mass Energy Transformation and Storage (VMETS) can be used [51–54] which can meet low energy density storage for solar cooling, heating, dehumidification etc. The solar energy in this study has been harnessed by using the ETC and the storage tank is also used to store energy. The VMETS

technology can transform the collected solar energy in to the chemical potential of the working fluid. The energy charging and discharging is dynamic which is either due to the external disturbances (solar radiation, cooling load and chilled water temperature) or internal disturbances (change in concentration and temperature of solution in storage tank), consequently changing the properties of working solution. In order to understand these variations a dynamic model is proposed which is based on following assumptions:

a. Heat capacities of all components in the system are infinitesimal. The resistance of flow in the fluid pipe connecting solution storage tank and solar collector is negligible.
b. The temperature of the metallic absorber plate and tube is uniform at the same section.
c. The power consumed by the solution pumps is negligible.
d. The solution storage tank is perfectly insulated, and the concentration and temperature is considered uniform in it.

The simulation studies were carried out for an area where the ambient temperature can reach upto 38 °C or more. The solar intensity and efficiency of the evacuated collector with a metallic absorber was calculated using a relation given by Duffie et al. [55] and ASHRAE [56] respectively. The various operating conditions were kept set for numerical simulation which includes the fixed AC load of 15.1 kW to be operated for 17 h in a day, indoor temperature is kept constant at 26 °C whereas the ambient temperature is varied between 29.5 °C and 38 °C, the absorber is cooled by water and condenser is cooled by either cooling air or cooling water, the solution charged into the system is having 50% concentration with a total weight of 800 kg, the effective length of the solar collector is 2 m which is installed at an angle of 30°. For the numerical simulation based on various operating conditions mentioned above, it is evident that at the beginning of the energy charging process the mass, concentration and temperature of the solution in the storage tank were found to be 763.1 kg, 52.42% and 48.4 °C (cooling air) or 41.9 °C (cooling water) respectively. The energy stored in the solution and water in storage tank is 231.5 MJ (cooling air) or 221.3 MJ (cooling water) and 36.9 MJ, respectively with the solar collector area of 66 m^2 (cooling air) or 62 m^2 (cooling water) having evacuated tubes 825 (cooling air) or 775 (cooling water) for the absorption system. The results obtained from the study also revealed that solar radiation energy on the solar collector was found to be 1719 MJ or 1623 MJ when condenser was cooled by air or water respectively. The energy received by solar collector was 782.3 MJ or 779.0 MJ, respectively and the average efficiency of the collector was found to be 45.5% or 48.0% respectively. The energy rejected from condenser is 610.9 MJ or 614.5 MJ and the energy rejected from the water storage tank is 24.6 MJ or 17.7 MJ.

The conclusions from the study indicate that under a typical operating condition, the COP of the new system could reach 0.7525 (cooling air) or 0.7555(cooling water) with the required solar collector area of 66 m^2 (cooling air) or 62 m^2 (cooling water) respectively. The non-uniformity between solar radiation and cooling demand can be removed by this method because it can use solar energy directly to drive cooling system instead of the use of hot water for cooling. They also provide a choice for condenser cooling either by cooling air or by cooling water. The air-cooled condenser elevates collector efficiency and hence requires more collector area where as water cooled condenser results in high efficiency but increases the system's complexity.

2.1.7. Solar driven double-effect absorption cycle for sub-zero temperatures

Many authors have discussed the results by distinguishing adsorption and absorption refrigeration systems on the performance as well as the exergy originated in the systems [57,58]. Different authors have also carried out the studies on the solar thermal absorption cooling systems based on the high temperature solar receivers [59,60]. A theoretical study of a solar driven double effect parallel flow absorption system with ammonia–lithium nitrate for sub-zero evaporation temperatures was carried out [61]. The long term performance is predicted for a slaughterhouse application in South Europe making use of a hot storage tank. The performance of a double-effect parallel flow ammonia based absorption system with lithium nitrate as absorbent has been investigated and is shown in Fig. 4. The system comprised of solar collectors, a double-effect absorption refrigeration system and a storage tank. The storage tank is added to the system in order to shift the excess thermal energy as much as possible from the period of high solar availabilities to the period of low solar availabilities. A parabolic trough solar collector, which can deliver high temperatures required to drive the double-effect absorption system, is considered because of its potential to harness more radiations in the morning as well as evening times.

During the higher solar radiation condition, the solar collector absorbs the incoming solar radiation, convert it into heat and transfer this heat to the fluid flowing through the collector. The collected solar energy is carried from the circulating fluid to the double-effect absorption system using a pump P1. The pump P2 kept in series with pump P1 which takes the circulating fluid from the double effect absorption system and the cycle completes. Two three way valves V1 and V2 are also provided in the system to take

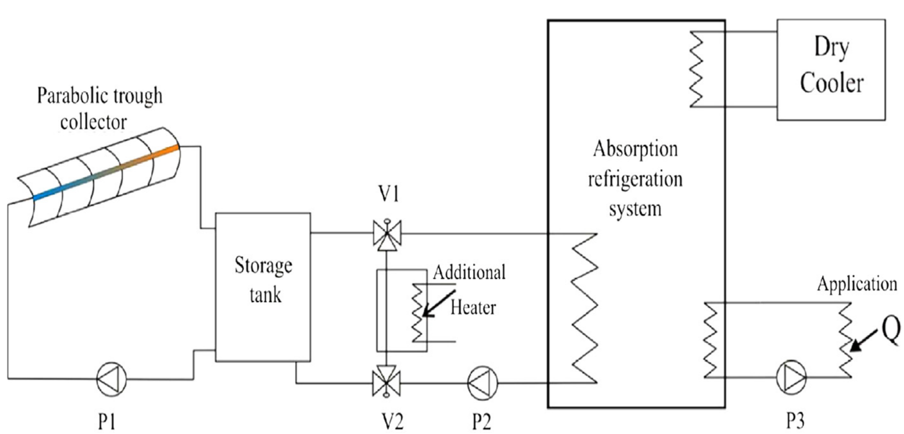

Fig. 4. Solar absorption system with a hot storage [61]. (The additional heater is only activated when the heating fluid temperature drops below the temperature required by the absorption system).

control of heat transfer fluid circulation eventually checking the temperature fluctuation in the system. The working of the system in the normal conditions (only with solar collector) will be P1–V1–P2–V2, with an additional heater in bye pass conditions. The three way valves remain fully open in the normal conditions. However, when the solar energy level falls below the threshold or there is a lesser energy storage in the storage tank, additional heater so provided starts operating. The system works in a sequence as P1–V1–P2–V2 considering heat transfer fluid circulates partially through the additional heater and this is how the temperature requirement in the generator can be checked by controlling the mass flow rate of circulating fluid. The analysis is carried out based on the fixed assumptions indicated as-(a) the pressure drops is considered negligible in pipelines and heat exchangers (b) the refrigerant at the outlet of the condenser and evaporator assumed to be as the saturated liquid and saturated vapour, respectively while the solution at the outlet of the absorber and generator is also in saturation state (c) the temperature of the solution at the outlet of the high pressure generator is assumed to be 5 K lower than the heating medium inlet temperature.

A modular computer simulation program was developed using object-oriented programming in C# language in Microsoft Visual Studio 2010. The main components of the absorption cooling system have been simulated separately based on the conservation of mass and energy as well as considering thermodynamic equilibrium. The thermodynamic properties for the NH_3–$LiNO_3$ solution have been obtained using equations developed by Farshi et al. [62] and REFPROP has been used for calculating ammonia properties [63].

The influence of the distribution ratio on the performance of the ammonia–lithium nitrate absorption refrigeration system has been investigated which indicate that the COP increases with the heat transfer fluid temperature. For each heat transfer fluid temperature, there is a distribution ratio that gives the maximum performance of the system. However for an increase in the heat transfer fluid temperature, the distribution ratio must decrease to obtain the highest performance of the system. The maximum amount of cold energy is produced when the distribution ratio is around 0.65.

The variation of the COP of the double-effect absorption refrigeration system is found to be rising with an increase of the driving temperature and decrease of the cooling water temperature. The attainable COP values were quite acceptable (0.5–0.95) considering the low temperature applications. The results obtained also indicate that power delivered in cloudy days even in the summer is significantly lower than for clear sky days so that the system can only operate in combination with a hot temperature buffer facility and/or with additional heater. Under the condition of solar radiation and 1000 m^2 solar collector area, the system can provide more than 50% of the heating energy required for the cooling load of a pork slaughterhouse with a peak load of 600 kW.

2.2. Ejector cooling systems

2.2.1. A solar ejector cooling system using refrigerant R141b

The different cooling technologies can use LiBr–water or ammonia–water as working fluids using a chemical absorption process. Huang et al. [64] carried a study for solar ejector cooing system using refrigerant R141b (1, 1-Dichloro-1-fluoroethane) as working fluid as shown in Fig. 5. An ejector cooling system (ECS) however, can be operated with low boiling point refrigerants as the working fluid and is proved to be the most promising device for solar cooling/refrigeration applications. Ejector cooling system utilizes the Rankine cycle along with the gas dynamic effect of ejector (thermal compression process) for cooling and is simple in designing with low installation cost. The only drawback of the ejector cooling system is its low COP usually lesser than 0.3 for a single-stage system [65–67] where as COP is found to around 0.4–0.6 for a single-stage absorption cooling system [3]. The system consists of components same as in conventional absorption system with an additional ejector in it and the specifications of the used ejector are listed in the Table 1 [64]. The solar collector (double glazed or flat plate) acts as the generator of the system and the mixing ejector is used here with the separator to recirculate the liquid working fluid through the solar collector to enhance the boiling heat transfer. The R141b is selected as the working fluid for the ejector cooling machine because of its high performance [66]. The energy collection efficiency of the collector is experimentally determined and results are experimentally verified according to the modified ANSI/ASHRAE 93-1986 Standard [68]. For the ejector performance analysis a dynamic model was used as proposed by Eames et al. [69] which include the secondary flow choking phenomena. The obtained results show that that the ejector cooling system designed for the study has a very high COP and can reach up to 0.5 for a generating temperature of 90 °C, condensing temperature of 28 °C and evaporating temperature of 8 °C. For the back pressure of an ejector exceeding the critical back pressure, the ejector performance drops dramatically at low evaporating temperature. The calculation of the collector efficiency was done using the collector inlet temperature at10 °C higher than the generating temperature of the ejector cooling system. Assuming no heat loss in the pipe, the results showed that the system would attain an optimum overall efficiency of around 0.22 at generation temperature of 95 °C and the evaporator temperature of 8 °C for solar radiation at 700 Wm^{-2}. The study also suggested that solar ejector cooling system can become a refrigeration system if it is operated at low evaporating temperature which becomes evident from the fact that solar ejector cooling attained an optimum COP value of around 0.12 for a

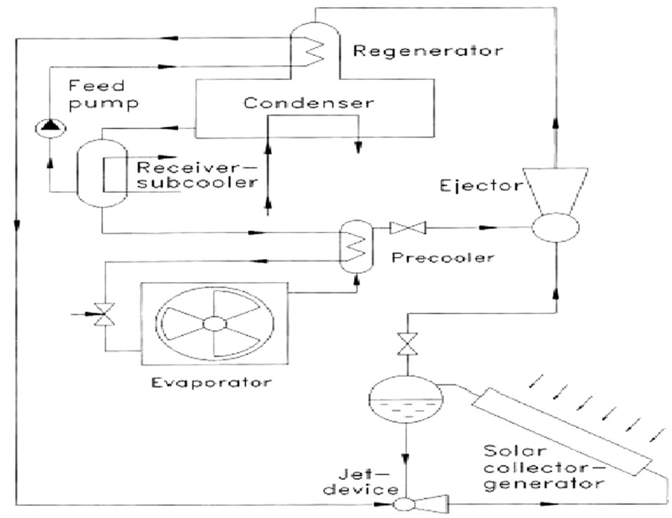

Fig. 5. Schematic diagram of solar ejector cooling system [64].

Table 1
Specifications of ejector [64].

Primary flow nozzle:	
Throat diameter (mm)	2.64
Exit diameter (mm)	4.50
Constant-area section:	Cylindrical type
Diameter (mm)	8.10
Diffuser angle (deg)	5
Distance between nozzle and constant area section (mm)	13

generation temperature of 102 °C and the evaporator temperature of −6 °C for solar radiation at 700 W^{-2}.

2.2.2. An exergy analysis of a solar-driven ejector refrigeration system

An exergy analysis of a solar-driven ejector refrigeration system to find the optimum operating conditions for the solar-driven ejector refrigeration system was done [70]. The analysis is performed by doing energy and exergy balances for the system. The schematic system diagram is shown in Fig. 6(a) consists of a solar collector subsystem and an ejector refrigeration subsystem. The major components in the refrigeration cycle are an ejector, a condenser, a generator, an evaporator, an expansion device and a pump. The vapour from the low temperature evaporator is sucked into the high velocity vapour stream in the ejector. The p–h diagram an ejector refrigeration system is also shown in the Fig. 6(b). The analysis of the solar collector sub system has been calculated using the equations already established by the author [71] and the analysis of the refrigeration subsystem is carried out by developing a model based on energy and momentum balance over the ejector. The pressure drop in the heat exchangers is neglected, and heat transfer is modelled by assuming typical temperature differences. The detail of the ejector is not included in the paper but the efficiency of the ejector is taken as 70% [72]. The mathematical model is implemented in the Engineering Equation Solver [73] and the properties for the refrigerant(s) are taken from NIST reference database REFPROP [74] for the analysis of the solar-driven ejector refrigeration system assuming the various fixed conditions i.e., (a) the incident solar radiation is 700 Wm^{-2} (b) double-glazed flat-plate solar collector is used (c) butane is used as the refrigerant for the refrigeration cycle (d) water is used as the heating medium between the solar collector and the generator (e) the outlet temperature of the solar collector is assumed to be 10 °C above the generating temperature (f) the cooling capacity is 5 kW (g) the ambient air temperature is 30 °C which also is used as the reference temperature for the analysis (h) the generating temperature is 90 °C (i) the condensing temperature is 37 °C (j) the evaporation temperature is 10 °C and, (k) the pump efficiency is assumed to be 25%. The results obtained indicate that the overall thermal efficiency at the generating temperature 90 °C is about 11%. About 58% of the solar energy supplied to the system is lost to the surroundings in the solar collector. A slightly higher overall efficiency is obtained at a generating temperature of about 80 °C due to lower heat losses in the solar collector. The coefficient of performance (COP) of the ejector cycle is 0.25 and the solar collector efficiency is about 48% for the conditions simulated. However, considering the components in the ejector refrigeration subsystem, the largest loss (32%) occurs in the ejector followed by the generator (22%) and the condenser (21%). The rest of the exergy loss occurs in the pump, the evaporator and the expansion valve. The exergy losses in the ejector are due to the friction losses of the flow inside the ejector nozzle because of the non-ideal adiabatic expansion in the nozzle. The performance of the ejector increases strongly at increased evaporator temperature. The second largest irreversibility in the refrigeration subsystem occurs in the generator. The major loss here is due to heat transfer over a finite temperature difference. The optimum generating temperature for the selected evaporation temperature of 10 °C is about 80 °C. Below this value the total irreversibilities increase and the optimum generator temperature depends on the evaporation temperature and other operating conditions such as condensing temperature, working fluid, and the efficiency of the ejector. At a given operating temperature, the evacuated-tube solar collector generates the lowest irreversibilities and gives the highest performance compared to the flat plate single glass cover and flat plate double glass cover solar collectors which were also analysed.

2.3. Adsorption cooling systems

2.3.1. Study on a solar heat driven dual-mode absorption chiller

Habib et al. [75] carried out a simulation study of a solar thermal driven dual-mode, four-bed silica gel–water adsorption chiller. The solar thermal collector data of Durgapur (23.48° N, 87.32° E), India has been used as the heat source to the chiller. For a driving source temperature above 60 °C, the chiller works as a single stage four-bed adsorption chiller; while it works as a two stage chiller when the driving source temperature falls below 60 °C [76–78]. Many authors have carried out the study on the different types of the solar cooling technologies indicating the feasibility of the solar cooling [79–85].

The performance simulation of dual-mode adsorption cooling system was carried out to analyze the changes in cooling capacity and COP with regeneration temperature and cooling water inlet temperature. The analysis was performed to overcome the drawback of the fluctuation of the temperature in the condenser and evaporator for the conventional two-bed adsorption chiller because the four-bed chiller helps in reducing the temperature fluctuations [86]. Also, Multi-bed mode helps to improve the recovery efficiency of low temperature heat sources like that obtained from simple flat plate solar collectors to useful cooling. The two-stage four-bed adsorption chiller comprises of six heat exchangers, namely a condenser, an evaporator and two pairs of

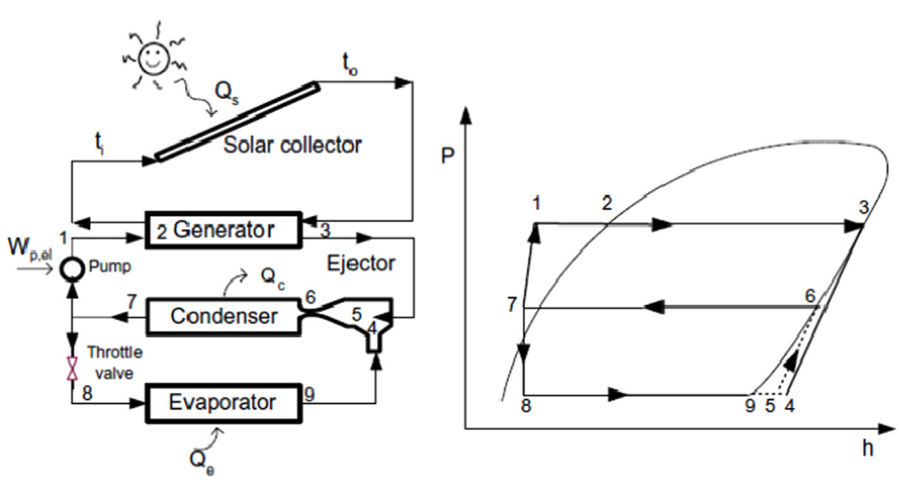

Fig. 6. (a) Ejector refrigeration cycle and (b) p–h diagram [70].

sorption elements and the cycle operates in four modes. The simulation of the major components such as four adsorber/desorber beds, the condenser and evaporator are solved numerically using energy balance equations based on the steady state conditions. To calculate the adsorption isotherms and kinetics, a series of subroutines of the IMSL Fortran Developer Studio software of the fifth order GDF (Gear's Differentiation Formulae) are used. The set of coupled ordinary differential equations for energy and mass balances were also solved. Double precision has been used with the tolerance value of 1×10^{-6}. A reliable data base has been used to calculate the value of solar thermal set up. The adsorption isotherm based on the modified Freundlich model have been used to estimate the equilibrium uptake of silica gel–water pair while the rate of adsorption or desorption is governed by the linear driving force kinetic model. The results obtained indicate that the monthly average hot water temperature of Durgapur for the dual-mode multi-bed adsorption chiller during the year 2011 obtained for the month of April, was around 90 °C, and the lowest hot water temperature was achieved in the month of August, which was about 48 °C which is in quite agreement of the main objective of using low temperature heat from solar energy as the driving source. However, for the constant chilled water inlet temperature of 14 °C, the two cases of the combination of the hot water and coolant are considered, (a) the hot water inlet temperature of 82 °C in combination with a coolant of temperature 30 °C and, (b) the hot water temperature of 60 °C along with a coolant at 30 °C and the results obtained indicate that adsorption chiller in single-stage operation is able to reach from transient to nearly steady state within three half cycles or 1900s, while the adsorption cycle in two-stage operation is able to reach steady state within 1800s. Also, in both operational modes, adsorbent cooling is clearly slower than desorption heating, which suggests that removal of adsorption heat is more problematic than addition of heat for desorption. It is also concluded form the study that the COP increases with the rise of regeneration temperature in both operation modes. The optimum values of COP achieved from two-stage, four-bed and single-stage, four-bed mode are around 0.2 and 0.45, respectively. Again it is noticeable that for both the single-stage and two-stage mode, cooling capacity increases as the cooling water temperature is lowered and is happening due to the fact that lower adsorption temperature promotes higher amount of refrigerant being adsorbed and desorbed during each cycle.

2.3.2. *Numerical modelling of a solar thermal cooling system under arid weather conditions*

A study based on TRNSYS simulation program for a 4.5 kW adsorption thermal cooling system installed in the office building was carried out [87]. The simulation program is carried for typical summer weather conditions in 2012 for Qatar which in turn used to analyze and optimize the performance of the solar-assisted cooling facility because of the high ambient temperature conditions leading to the high electricity consumption in air-conditioning units. The study utilized the local meteorological data to assess the solar cooling performance based on the cooling demand. The optimisation process is carried out using the Typical Meteorological Year (TMY) data bank from TRNSYS for Doha city. The weather data was logged onto the New Energy Lab system to register global radiation, wind speed and ambient air temperature in 1 min interval over the period of 20 days from 22/5/2012 to 10/6/2012. The different operating parameters were analyzed based on the simulation study and consequently the various performance parameters were also analyzed as discussed below. The analyzed performance parameters were used to optimize the specifications of cooling system providing the cooling load of the 37 m^2 in office space in the Doha city using the local meteorological data. The optimum values so obtained from the simulation study for the designed system are summarized in the Table 2 [87].

The whole cooling system is divided into three main circuits called primary, secondary and tertiary circuits meant for different applications. Solar energy is harvested from the evacuated tube collector and circulated in the primary water circuit to maintain its hot water temperature. The harvested energy then transferred to the hot water storage tank through the secondary water loop using a heat exchanger. In the tertiary water circuit, pump is used to deliver hot water to the adsorption system when cooling demand is required at specific operating hours. Auxiliary heater is also provided to boost the hot water temperature when it falls below the threshold value. The mathematical model for the evacuated tube collector efficiency is calculated using Hottel Whillier equation [88,89]. The flow rate in the primary water circuit is maintained between 0.028 kg/s and 0.083 kg/s when the collector is gaining energy from the Sun. A stratified storage tank with 6 nodes is modelled in the simulation. The overall tank loss coefficient is assumed to be 3.6 kJ/hm^{2}K. The studied volume ranged between 0.1 m^3 and 2 m^3. An auxiliary heater having heating capacity of 10 kW is connected to maintain the desired hot water temperature and is kept at set point temperature of 70 °C. A commercial adsorption chiller InvenSor model LCT 10 is used in this study to assess the cooling load of the 37 m^2 office space and according to ASHRAE standards [90] at an outdoor temperature of 45 °C; the thermal comfort load for indoor temperature of 24 °C is about 4.5 kW.

The simulation study of the effect of the evacuated tube collector angles with its daily solar energy gain revealed that the daily solar energy gain is influenced by the orientation of the evacuated tube collector. The simulation was conducted on total aperture area of 23.4 m^2 and the range of studied angle was between10° and 34° which suggested that an optimum angle in Doha weather environment is 24° for evacuated tube collector. It is however, also concluded that it is crucial to select an optimum tank size for solar-assisted cooling system. An oversized water tank often requires more auxiliary power than a smaller water tank. The study of the effect of water storage volume on the system performance revealed that smaller tank volume has higher energy gain than a larger tank. When the tank size increased over 0.4 m^3, energy gain from the solar collector started to decrease and is happening due to excess heat capacities available in larger storage tank. The analysis carried out for different tank volumes revealed that for the tank volume of 0.3 m^3, the water outlet temperature rises up to 66 °C within 1 h, where as for larger tank volume of 0.8 m^3 and 1.4 m^3 the water temperatures can only reach up to 59 °C and 52 °C, respectively for same time period. With the tank volume of 0.3 m^3, the outlet water temperate plunged slight below 68 °C in the second hour because the hot water demand was higher than the water storage can deliver. Contradictory, in larger water storage tanks, not significant water temperature drop in the second hour is observed in this study.

Table 2
Optimum values of design parameters [87].

Parameters	Type/value
Collector type	Evacuated tube
Cooling capacity	4.5 kW
Collector area	23.4 m^2
Collector slope	24°
Water storage tank volume	0.3 m^3
Heat exchanger flow arrangement	Counter-flow
Heat exchanger effectiveness	0.75
Secondary water circuit flow rate	0.139 kg/s

Therefore, the optimum tank size needed for the system is found to be 0.3 m^3. The effect of secondary water circuit flow rate and the heat exchanger effectiveness is also analyzed in the study. A counter-flow heat exchanger has been selected in the study because of its high rate of heat transfer compared to a parallel flow heat exchanger. The four heat exchanger effectiveness was analyzed and the results obtained indicate that the water energy gain increases with the heat exchanger effectiveness. It has also been concluded that an effectiveness of 0.75 is found to be suitable for the system as it is readily available in the heat exchanger market. The secondary water circuit flow rates in the range from 0.028 kg/s–0.458 kg/s have also been studied which revealed that the energy gain in the tank reaches its peak at flow rate of 0.139 kg/s for all four different heat exchanger effectiveness. The energy gain decreases gradually when flow rate increased over 0.139 kg/s and this could be due to the solar collector has reached it maximum output energy. The effect of solar collector area and auxiliary heating demand based on simulation study revealed that that greater the solar collector area, the less energy is required from auxiliary heater. However, an optimum collector area also depends on the economical value such as initial investment and operational costs. The annual running cost was also calculated at the rate of 0.06 $/kWh which reveals that economically viable collector area for the system is 23.4 m^2. The economical analysis so carried out was based on the assumption that 30% of the initial cost to be paid in advance and the remaining amount is to be paid in instalments over the period of 10 years.

As a concluding remark, it is worthy to mention that the solar-assisted adsorption cooling systems can reduce the power demand from national grid while providing same amount of cooling as conventional compression cooling. The power consumption of a same cooling capacity compression system with the coefficient of performance of 2.6 consumed 15.3 kW h of energy in 9 h of operation whereas solar assisted adsorption cooling system only consumed 10.4 kW h of energy thus cut down the electric energy demand by 47%.

2.3.3. A stand alone solar adsorption refrigerator for humanitarian aid

The traditional vapour compression refrigeration system driven by internal combustion engines involves a series of drawbacks and problems concerning fuel supply and maintenance. Thermally-driven adsorption cooling systems have been extensively studied and can be considered as a viable alternative to traditional electric-driven vapour compression systems [91] and the technology sounds particularly attractive when an high amount of low temperature heat is available, such as solar energy [92]. Santori et al. [93] developed and analyzed the solar driven ice maker operating with the activated carbon/methanol adsorption pair as supporting humanitarian aid actions for vaccines storage. Such analysis was carried out for the countries usually have not a fully-developed electrical grid. The work done was based on the development of a new solar-powered adsorption ice maker using activated carbon (adsorbent) and methanol (adsorbate) as working pair. The prototype was designed on the basis of the results of mathematical models developed and optimized in previous works [94–96] and the operational phase is shown in Fig. 7. The prototype has been developed using ideas from previous existing activities with the aim to study the influence of some different, modifiable aspects (i.e. heat transfer surfaces areas, type of adsorbent, adsorbent mass and adsorbent grain size) on the overall performance. The machine operates with a 24 h intermittent cycle and consists of the components such as- a solar collector in which the adsorbent (activated carbon) is integrated, an air-cooled condenser for the relative adsorbate (methanol) phase transition and an evaporator placed inside an insulated box where cooling is produced.

The most important part of the system is the adsorbent bed where the pressure gradient generated because of the thermal conditions, helps in the vapour movement through the other components. Adsorbent bed consists of a tube bundle made of 10 pipes loaded with 20 kg of adsorbent. Activated carbon SRD 1352/3 with a grain size between 0.6 and 1.7 mm (origin Coconut Shell, manufactured by Chemviron Carbons Ltd.) [97] has been suitably selected to work in pair with methanol which guarantee a sufficient mass transport through the tube length. The adsorbent bed is integrated with a flat-type solar collector with an exposed area of 1.2 m^2. The advantage of this configuration is the compactness of the component which in turn helps in the reduction of the heat transfer resistances from solar collector to the activated carbon. The system is equipped with pressure (piezoresistive) and temperature sensors (thermocouple type T, class 1), with an accuracy respectively of ± 2 mbar and ± 0.5 °C respectively. The most relevant operational variables during the adsorption desorption process ere also monitored. A Pyranometer for the measurement of the global radiation is also installed on the same plane

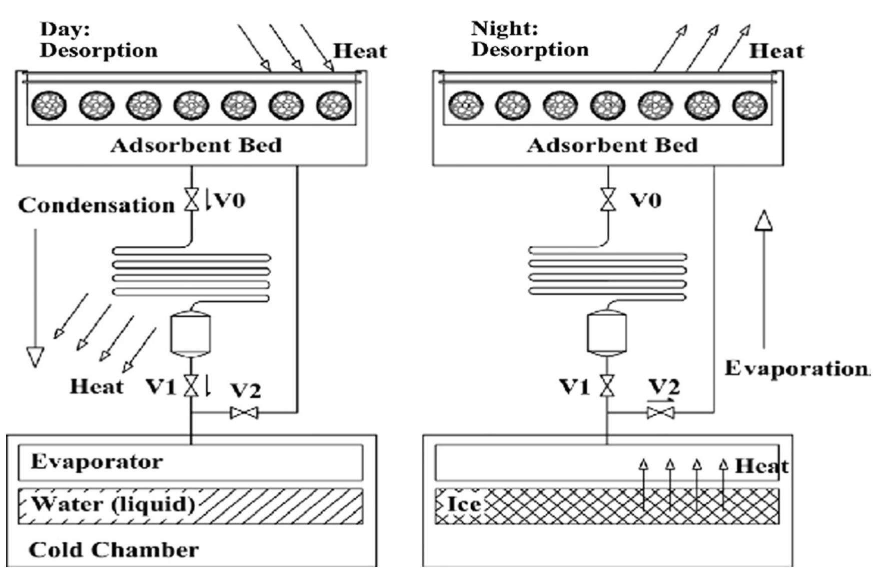

Fig. 7. Operational phases of an intermittent solar adsorption ice maker [93].

of the solar collector. The main features of the designed prototype are mentioned in the Table 3 [93]. The installed prototype has been tested for two typical days, i.e., on 11 February 2013 (winter season) and on 20 March 2013 (spring season). During the diurnal time, the solar radiation is effectively captured which allow to attain high temperatures on the adsorbent grains in both the days. It has also been observed that carbon temperature rises steadily until achieve its maximum, near to 93 °C and 85 °C respectively on February and March. During the night time (adsorption phase) the external temperature permits to cool down the adsorbent bed promoting the methanol adsorption and the consequently produces a useful cooling effect in the cold box. The evaporator temperature (measured on the external surface of the heat exchanger) decreases respectively to −13.6 °C on 11 February and −5.3 °C on 20 March allowing the cooling of the air closed in the box and the subsequent ice production. These minimal temperature values were attained in about 5 h after the beginning of the adsorption phase. The solar Coefficient of Performance (COPs) obtained was 0.08 on 11 February and of 0.063 on 20 March. The results obtained also revealed that diurnal ambient temperature is another important operational parameter as it directly influence the methanol condensation rate stating that operation of the system at slightly higher ambient temperature than the design ambient temperature, coupled with a slight decrease in the desorption temperatures, leads to a plentiful decrease in the machine performance.

2.4. Study of the solar assisted vapour compression absorption cascaded air-conditioning system

Chinnappa et al. [98] carried out the study of a solar assisted vapour compression absorption cascaded air-conditioning system using ammonia–water to overcome the global environment challenges. The water cooled technique has been replaced with air-cooled heat exchangers. The system could be operated in three modes-(a) use of conventional system only (b) use of the hybrid system only, and (c) use of each (a) and (b) intermittently. The ETC based solar collector is used to provide the hot water (85–95 °C) which is mounted at the slope of 10°. Seven parallel circuits of three collectors in series have been used. The aperture area of each collector is 1900 mm × 985 mm. The absorber is made up of mild steel coated black to ensure good absorptivity and emissivity. The hot water from the collector is stored in the hot water storage tank which can be pumped as an energy input to the generator of the absorption system. AD590 sensors, Bourdon gauges and magnetic flow metres were used for temperature measurements, pressure indication and flow measurements respectively. The energy consumed by compressors, fans and heaters were monitored using energy metres. The solution circuit has three possibilities of operation, first in case of hybrid system, when the ammonia tank is full, the solution tank contains the weak solution and ammonia after expansion passes through the cascade evaporator, weak solution is drawn from the solution tank and is absorbed also. The flow of solution is enabled through the pump. This mode of operation is carried when there occurs no generation of ammonia such as during night or cloudy days when the air-conditioning system needs to be operated, the second option follows. When the solution tank contains the strong solution, it can be pumped through the generation circuit consisting of a solution heat exchanger, a boiler, a phase separator, and a solution float tank. The ammonia vapours are condensed in an air-cooled condenser and again sent to the ammonia storage tank where the concentration of ammonia gradually becomes the weak. This option is useful over the weekends and when no load exists on the R-22 system; but solar energy is quite frequently available for generation purpose. Thermodynamically, the best performance of a refrigeration system is obtained when the difference between the condensing and the evaporating temperatures is about 30 °C. The results obtained indicate that when the ambient temperature is varied from 27 °C to 32 °C, the temperature variation in the absorption and compression systems depend on the intensity of solar insolation in the solar collector as well as carryover of ammonia in the storage tank. The study also revealed that an insolation of 507 W/m^2 is required to generate ammonia and when it is less than 345 W/m^2 no generation is possible. The carryover of ammonia is governed by the level of insolation and the cooling demand. The condensation of R-22 occurs at about 27 °C, while the evaporating temperatures of ammonia lie in the range of 21–25 °C. For the test duration, the solar collector efficiency was found in the range of 0.43–0.50. The COP of absorption system found to lie in the range of 0.59–0.72. The compressor system yielded a COP of 2.55 with an actual power consumption of 4.36 kW. The power consumption of the hybrid compressor was 2.2 kW and the overall hybrid COP was about 5. The results obtained also indicate the energy saving when the hybrid system is functional besides certain indirect benefits which includes reduction in compressor load, increases the compressor valve performance and smaller discharge superheats. The results obtained also revealed that when hot water temperature is of the order of 95 °C, the generation of ammonia is possible and when the temperature falls below 80 °C no ammonia generation takes place. The results obtained from the study also points that the exergetic efficiency of the compression system is large when operating temperature differential is of the order of 30 °C in comparison to temperature differential of 45 °C.

Table 3
Main features of an ice maker [93]. (Designed prototype).

Solar collector	
Exposed area	1.2 m^2
Selective surface	Sol Max Foil ($\alpha = 95-99\%$; $\varepsilon = 4-10\%$)
Tube bundle	5 tubes DN 60 × 1.73 m + 5 tubes DN 60 × 1.63 m
Total surface	3.7 m^2
Absorbent material	SRD1352/3 Chemviron Carbons Ltd.
Absorbent mass	20 kg
Grain size	0.6–1.7 mm
Condenser	Air cooled, 7finned tubes DN 16, length 1 m Cylindrical receiver (DN 100, 6.5 l)
Total surface	4.08 m^2
Evaporator	7finned tubes DN 25, length 1 m in two interconnected levels
Total surface	18.45 m^2
Insulated box	
Internal volume	1000 × 640 × 500 mm = 0.32 m^2
External volume	1.3 m^3
Insulation material	Polyurethane foam

2.5. Compressors driven by thermal solar energy: entropy generated, exergy destroyed and exergetic efficiency

Izquerdo et al. [99] carried out a study of entropy generation, exergy destroyed and the exergetic efficiency of Lithium–Bromide absorption thermal compressors for a single effect and double effect systems driven by solar thermal collectors. Two different applications of the system, which is air cooled and water cooled have been analyzed and compared. The water cooled compressors works with pressure and temperature lower than air cooled compressors. A single effect absorption system is shown in Fig. 8 consists of components same as that of conventional absorption

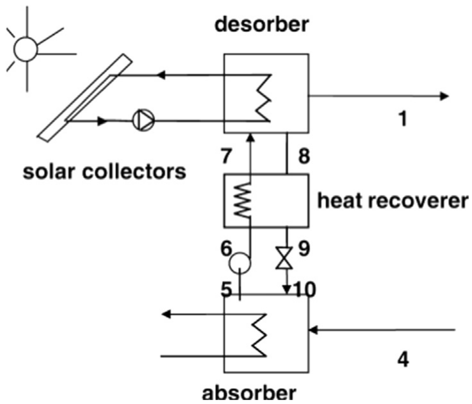

Fig. 8. Single effect thermal compressor fed by a thermal collector [99].

system where generator works as desorber and the double-effect system consists of similar components with an additional desorber, a heat recoverer and throttling device. The component desorber, absorber, heat recoverer and throttle valve absorbs vapours at low temperature and pressure and deliver them at high temperature and pressure at condenser inlet. Water is commonly used as heat transfer medium to remove the heat from the thermal compressors through absorber. The thermodynamic properties and working conditions of the working fluid for single and double effect have been used as described by Duhring, Merkel and Mollier diagrams. The entropy of the solution was obtained from Feuerecker et al. [100] and that of refrigerant were obtained from Mollier diagrams. The results obtained showed that the temperature and pressure of air-cooled thermal compressors are higher but remain very close to the water cooled temperature and pressure values. The maximum boiling temperature of the desorber of single-effect compressor reaches up to 110 °C at 0.12 bars while that of double-effect compressor reaches up to 180 °C at 0.5 bars. The thermal compressor (operated by solar collectors) absorbs the low pressure refrigerant vapours coming from the evaporator and produces superheated refrigerant vapours. The temperature of about 120 °C is required for the compressors and can be achieved from solar thermal collectors of low concentration or vacuum pipe solar converters (with or without concentrators also known as CPC's). The efficiency of the vacuum tubes can be calculated from the expressions provided by VIESMANN and for CPC's the efficiency can be calculated using the relation given by SOLEL CPC 2000. The energy and mass balance equations in the different components have been used to calculate the entropy production, entropy destruction of different components as well as exergy efficiency of the single-effect as well as double-effect system. The double effect thermal compressors, requires discharge temperature of 180 °C when solar thermal compressor is cooled by air at round 40 °C. If it has to be cooled with water, the discharge temperature required is 150 °C. The air-cooled systems don't use cooling towers and hence are economical as well as attractive. However, the flat plate-collectors are not able to generate temperature of about 200 °C and for the reason the parabolic concentrators, with concentration ranging from 20 to 40 or last generation vacuum tube collectors (VAC 2008 type of SOLEL) are used to provide temperature of about 200 °C. The results obtained from simulation study indicate that the air-cooled compressors generate more entropy in comparison to water-cooled compressors. The double-effect system generates less entropy in comparison to single-effect system. The double-effect compressors in both air-cooled and water-cooled cases has higher exergetic efficiency than single effect systems.

2.6. Comparative study of different types of solar cooling systems for buildings in sub tropical city

Solar cooling is a sustainable mean to provide air-conditioning and refrigeration and is also a feasible way to replace electric cooling machines. Therefore, the different types of solar cooling systems included in the study were solar electric compression refrigeration, solar mechanical compression refrigeration, solar absorption refrigeration, solar adsorption refrigeration and solar solid desiccant cooling [2]. In this study, each type of the solar cooling systems was designed to serve a common typical office which was single storey with floor area of 200 m^2. The study revealed that flat plate collectors and evacuated tube collectors are the common types of solar collectors used for solar thermal applications but parabolic collectors are also becoming popular these days.

In case of solar electric compression refrigeration system, the various systems were built using TRNSYS and TESS [101,102]. The major equipment included are PV panels, direct current (DC) motor, power regulator, vapour compression chiller, chilled water pump, cooling tower, condenser water pump, air handling unit (AHU). The modelling of DC motor was done as suggested by Hughes [103] and that of condenser and evaporator as suggested by Lee [104]. The compressor estimation was based on a technique suggested by Jin and Spitler [105]. An auxiliary system was also used to draw power during bad weather conditions.

The solar mechanical compression refrigeration system consists of solar collectors, hot water pump, hot water tank, auxiliary heater, regenerative water pump, heat engine using Rankine cycle, vapour compression chiller, chilled water pump, cooling tower, condenser water pump and air handling unit. Modelling approach of the Rankine cycle and turbine was based on models proposed by Putten and Colonna [106] and Dixon [107] respectively. The modelling methods of remaining components were done in the same way as discussed above. The minimum driving temperature required for the system is 82 °C and if it is lesser there would only be heat loss and no cooling effect is produced. An auxiliary heater with fixed temperature of 90 °C is used to draw the energy during an insufficient solar energy supply. The solar absorption refrigeration system as shown in Fig. 9 consists of solar collectors, hot water pump, hot water tank, auxiliary heater, regenerative water pump, absorption chiller, chilled water pump, cooling tower, condenser water pump and air handling unit (AHU). The enthalpy of lithium bromide (LiBr) solution was used as given by Florides et al. [108] and the saturated vapour pressure of LiBr solution was taken from the Patek and Klomfar [109]. The various properties of refrigerant (water) were determined according to Florides et al. and Zhang et al. [110]. For the cycle operation, the utilization of auxiliary heating should be kept as low as possible by adopting a part load control of chiller.

The simulation model of solar adsorption refrigeration system was designed and is shown in Fig. 10 which was similar to that of the solar absorption refrigeration except an adsorption chiller was used instead. For adsorption refrigeration, an adsorption pair is also used which includes silica gel (adsorbent) and water (adsorbate). In a working cycle, two working chambers were used, one of the chambers was used for adsorption while another for desorption. Model development of the solar adsorption refrigeration cycle was based on the validated model of Cho and Kim [111].

The solar solid desiccant cooling system consists of solar collectors, hot water pump, hot water tank, auxiliary heater, regenerative water pump, regenerative heating coils, desiccant wheel, heat wheel, evaporative coolers, supply air fan and exhaust air fan. The model of desiccant wheel was developed based on the validated results as given by Zhang et al. [110]. Different from other four types of solar cooling systems, the solar desiccant

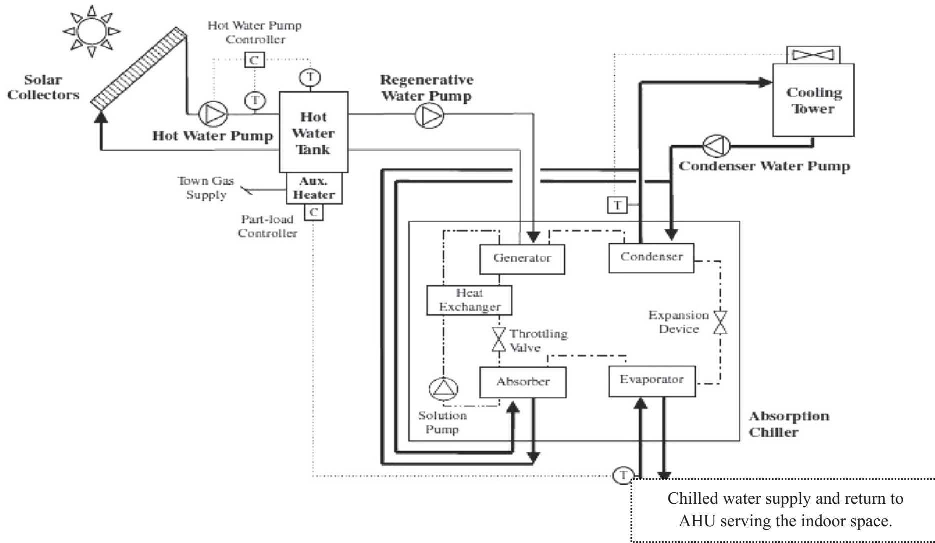

Fig. 9. Schematic diagram of solar absorption refrigeration [2].

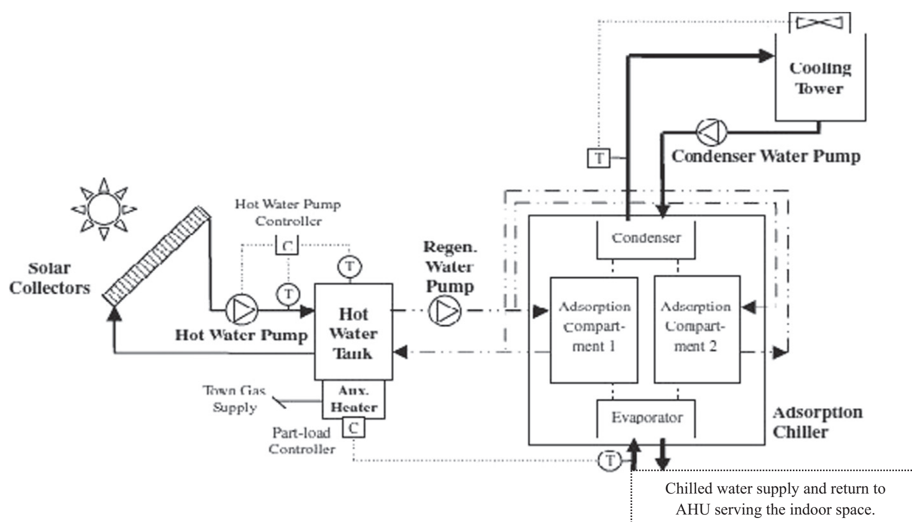

Fig. 10. Schematic diagram of solar adsorption refrigeration [2].

cooling would directly treat the fresh air for air-conditioning purpose. The desiccant wheel was sized according to the designed regeneration temperature of 85 °C.

In order to have a thorough comparative study of different solar cooling systems, the types of installation strategies and solar collectors were also studied which includes open space roof mounting at an inclination of local latitude and building-integrated mounting on the facades. These two installation strategies for solar electric compression refrigeration system and solar absorption refrigeration system were also studied. The performance is evaluated on the year round basis indicating that solar collectors integrated to building facades can reduce transmission heat gains by 12.7% and 9.2% for building integrated solar electric compression refrigeration and solar absorption refrigeration respectively as compared to roof mounting strategy. The year-round performances of the solar absorption refrigeration system for different types of solar collectors, including the flat plate collectors (used as the base line of comparison), evacuated tubes and parabolic concentrators were also studied which revealed that both the evacuated tubes and parabolic concentrators had higher solar fraction and solar thermal gain, and lower primary energy consumption. However, since the parabolic concentrator only responded to the direct solar irradiation, its performance was found to be lower than that of the evacuated tubes despite a higher collector temperature could be achieved. In terms of primary energy consumption, the solar electric compression refrigeration was found to be the best and solar mechanical compression refrigeration was found to be the worst. In fact, the solar electric compression refrigeration and solar absorption with evacuated tubes had comparable performances. However, the primary energy consumption of the solar absorption refrigeration system was found to be 10.8% higher in comparison to other systems. The solar adsorption refrigeration stood third in view of the primary energy consumption. Although it's primary energy consumption was about double to that of solar electric compression refrigeration or solar absorption refrigeration and the various researches is on the move for new and more effective adsorbent–adsorbate pairs [112–114]. On the other hand, there are emerging configurations for enhancing the effectiveness of adsorption–desorption process [26,115,116]. The average COP value for the summer showed that solar solid desiccant cooling is the best and solar mechanical compression refrigeration is the worst. The results obtained from the study also showed that the building-integrated approach could reduce the space cooling load by about

Table 4
Comparative tabular representations for different type of solar cooling systems with methodology adopted and the results obtained.

S. No	Cooling option	Methodology adopted	Results
1.	Absorption cooling systems		
	(i). Solar powered absorption cooling machine [18]	A simulation study based on model program containing physical properties of all materials and working fluid pairs was carried out to calculate the temperature and concentration in the desorber/absorber.	The COP obtained is quite less but with certain modifications in the system there can be an improvement in the performance.
	(ii). New absorption cycle using solar energy [22]	Simulation study based on various assumptions has been used to calculate the different performance parameters.	The COP calculated in the study is found to be quite higher than the conventional system because of higher rate of energy utilization in the new system. It also becomes evident from the obtained results that the condenser heat load is also important parameter as it clearly identifies the operational cost of the system.
	(iii). Optimum hot water temperature for absorption solar cooling [24]	An algorithm based on characteristic equation model was considered that could yield optimum hot water temperature needed for the generator.	The analysis stated that the there exists a hot water temperature below which the machine is not capable of cooling but consumes heat because of internal losses. The study also revealed that to maintain maximum generator temperature corresponding to maximum solar coefficient of performance along with time requires control of the mass flow rate through the collector.
	(iv). First and Second law analysis of a solar energy based power and refrigeration cycle [31]	A simulation is carried out with the optimization program GRG2 which uses the generalized reduced gradient (GRG) algorithm to optimize the cycle [32].	The results obtained indicate that irreversibility is mainly due to the absorber, recovery heat exchanger and rectifier. The highest irreversibility is found to be in absorber which is 44%. The irreversibility in rectifier and SHE is found to be 16 and 24% respectively [34] which could be reduced by using different refrigerant mixture and also through better matching of heat source temperature.
	(v). Solar collector driven water heating and absorption cooling plant [35]	The numerical method based on 4th–5th order Runge–Kutta equations integrated on adaptive time step [47] has been used and the net heat transfer rate for the collector surface is calculated as suggested by Buzelin et al. [46] which were based on energy balance.	The outcomes suggested that maximum second law efficiency is sharp, stressing its practical contribution in designing of such systems to be commercially competitive. The fluid capacity rate of thermal collector is also found to be of the highest importance as it is an important finding for the purpose of system scalability. The optimization based on mass flow rate is also carried out.
	(vi). Solar powered absorption refrigeration system with advanced energy storage technology [48]	The simulation studies based on considering various conditions pre set were carried out. The solar intensity and efficiency of the evacuated collector was calculated using a relation given by Duffie et al. and ASHRAE respectively [55,56].	The COP of the system can reach 0.7525 (cooling air) or 0.7555(cooling water), the required solar collector is 66 m^2 (cooling air) or 62 m^2 (cooling water) as suggested by the study. The air-cooled condenser elevates collector efficiency and hence requires more area where as water cooled condenser results in high efficiency but increases the system's complexity [48].
	(vii). Solar driven double-effect absorption cycles [61]	A modular simulation program using C# language in Microsoft Visual Studio 2010 was developed. The parabolic trough collectors have been used in the study.	A study of a solar energy driven double effect parallel flow absorption system for sub-zero evaporation temperatures revealed that COP lie in the range of 0.5–0.95 which is quite acceptable for low temperature applications. The performance of the system can also be improved, if system can operate in combination with a hot temperature buffer facility and/or with additional heater [61].
2.	Ejector cooling systems- (i). Solar ejector cooing system [23]	1-D gas dynamic model has been developed which also includes the choking phenomenon in the secondary flow [27]. The ejector performance has been calculated using a dynamic model [28]. Solar collector (double glazed or flat plate) acts as the generator of the system.	The system will achieve an optimum overall efficiency around 0.22 at generation temperature of 95 °C and the evaporator temperature of 8 °C for solar radiation at 700 Wm^{-2}. The system however, can also be used as a refrigeration system if it is operated at low evaporating temperature [23].
	(ii). Exergy analysis of a solar-driven ejector refrigeration system [85]	The analysis of the solar collector system was done using the equations [86] and the analysis of the refrigeration system is carried out by developing a model based on energy and momentum balance over the ejector. The double-glazed flat-plate solar collector has been used in the analysis.	The exergy analysis of the ejector refrigeration system revealed that the largest loss (32%) occurs in the ejector followed by the generator (22%) and the condenser (21%). Although, the performance of the ejector increases strongly at increased evaporator temperature. The optimum generating temperature for the selected evaporation temperature of 10 °C is about 80 °C [85].
3.	Adsorption cooling systems		
	(i). Solar heat driven dual-mode adsorption chiller [75].	A series of subroutines of the IMSL Fortran Developer Studio software of the fifth order GDF (Gear's Differentiation Formulae) were used to calculate the adsorption isotherms and kinetics. The set of coupled ordinary differential equations for energy and mass balances are solved.	The study of a solar thermal driven dual-mode, four-bed adsorption chiller revealed that optimum values of COP achieved from two-stage, four-bed and single-stage, four-bed mode are around 0.2 and 0.45, respectively.
	(ii). Solar thermal cooling system under arid weather conditions [87]	A simulation study using TRNSYS software for a 4.5 kW adsorption thermal cooling system based on the local meteorological data to assess the solar cooling performance was carried out. The evacuated tube collector was used and efficiency is calculated using Hottel-Whillier equation [88,89].	The study helps in determining the optimized system configuration for the Doha city. However, the study also states that solar-assisted adsorption cooling systems can reduce the power demand from national grid while providing same amount of cooling as conventional compression cooling.
	(iii). A stand alone solar adsorption refrigerator system [93]	The prototype was designed on the basis of the results of a mathematical models developed and optimized in previous works [94–96]	The solar Coefficient of Performance (COPs) of 0.08 on11 February and of 0.063 on 20 March was obtained. The diurnal ambient temperature was found to be an important operational parameter as operation at slightly higher ambient temperature than the

			design ambient temperature, coupled with a decreasing desorption temperature, leads to a plentiful decrease in the performance of the machine.
4.	Solar assisted vapour compression absorption cascaded system [98]	Data has been logged using a Cleveland Personal computer and the necessary software for switching on/off the entire setup as well as several subsystems has also been developed. The ETC based solar collector is used to provide the hot water in the range of 85–95 °C.	The collector efficiency was found in the range of 0.43–0.50 while COP of absorption system is found to lie in the range of 0.59–0.72. The exergetic efficiency of the compression system is calculated to be the maximum.
5.	Compressors driven by solar thermal energy [99]	The thermodynamic properties and working conditions of the working fluid for single and double effect have been considered as described by Duhring, Merkel and Mollier diagrams. The entropy of the solution was obtained from Feuerecker et al. [100] and that of refrigerant was obtained from Mollier diagrams. The thermal collectors of low concentration or vacuum pipe solar converters (with or without concentrators also known as CPC's) are also used to harness solar energy.	The application of thermal compressors for a single effect and double effect systems revealed that that the air-cooled compressors generate more entropy in comparison to water-cooled compressors. The double-effect compressors in both air-cooled and water-cooled cases has higher exergetic efficiency than single effect system.
6.	Comparative study of different types of solar cooling systems [2]	Different simulation technologies based on the different tools, softwares and already established techniques were used in the analysis [26,101–116]. The flat plate collectors, evacuated tube collectors and parabolic trough collectors are used in the study for the different systems depending on the requirement.	The comparative performance study of different solar cooling systems on the basis of the primary energy consumption is of the order as mentioned below- **_Solar electric compression refrigeration > Solar absorption refrigeration > Solar adsorption refrigeration > Solar solid desiccant cooling > Solar mechanical compression refrigeration_** It has also been concluded that the solar electric compression refrigeration and solar absorption refrigeration were the two types of cooling systems that could have energy saving potential when compared to conventional vapour compression systems. The year-round energy savings would be from 15.6% to 48.3% for air cooled refrigeration, while 8.0–43.7% for the water cooled refrigeration [2].

10% but the performance was found to be worse. For the choice of solar collectors, the parabolic concentrators had the primary energy consumption only slightly better than the flat plate collectors by 7.3% for the absorption refrigeration, but worse than the evacuated tubes by 36.5%. For the conventional electric-driven air-cooled and water-cooled vapour compression chiller plants included in the comparison, the solar electric compression refrigeration and solar absorption refrigeration (with evacuated tubes or flat plate collectors) were the two types of solar cooling systems that could have attractive energy saving potential. The continual advancement of the adsorption pair and chiller configuration would enhance the use of its distinctive merit of low driving temperature.

3. Discussion of results

The study of the different types of the cooling systems operated with solar energy has been carried out and the results indicate that the solar energy is a better option as an energy source to run cooling systems with an only drawback is the inconsistency giving lesser COP. The solar energy is readily available and needs to be harnessed in a better way to fetch out the maximum output. The test and simulation of the solar power cooling absorption machine has been carried out for dry absorption cooling system with a collector integrated reactor. The results indicate that the ammonia with combinations other than water can be used for cooling applications. In this case strontium chloride has been used in the cooling application. The results indicate that the performance of the system is very low but with certain modifications in the system the performance of system can be improved as indicated when results obtained are validated with the work done by other authors. The results obtained from the study of the solar assisted vapour compression absorption cascaded air conditioning system revealed that for the generation of ammonia for the whole hour, an insolation of 507 W/m^2 is required, when it is less than 345 W/m^2 no generation is possible. Also, when the system is analyzed in the duration from 0930 hours to 1500 hours on the test day, the solar collection efficiency was in the range of 0.43–0.50. The absorption system coefficient of performance (COP) ranged from 0.59–0.72.

Another study revealed that the compressor alone system yielded a COP of 2.55 with an actual power consumption of 4.36 kW where as the power consumption of the hybrid compressor was 2.2 kW and the overall hybrid COP was about 5. In addition to the above, exergetic efficiencies of compression systems are larger when the operating temperature differentials are of the order of 30 °C when compared with the operating temperature differentials of around 45 °C. The information gathered from the experimental investigation reveals that generation of ammonia is faster if the hot water temperature at the inlet to boiler is about 95 °C, and virtually no generation is possible when this temperature falls below 80 °C. These conditions are realizable with the use of good quality solar flat-plate collectors with selective coatings and regular cleaning of the glass surface. The study of another refrigeration cycle driven by heat and electricity revealed that minimum temperature obtained at evaporator and the COP is the main defining parameters for refrigeration. The evaporating temperature should be higher than minimum temperature to produce the positive refrigeration effect where as COP calculated found to be quite higher than the conventional system because of higher rate of energy utilization in the new system. Besides the performance factor stated above, condenser heat load is also an important parameter as it identifies the operational cost of the system. The first and second law analysis of a refrigeration cycle using solar energy revealed that the heat transfer rates in a cycle are the

function of the heat source temperature and increases with an increase in the heat source temperature, which eventually increases the irreversibility. The results also indicate that irreversibility in the cycles is mainly due to the absorber, recovery heat exchanger and rectifier and such losses could be reduced by using different refrigerant mixture and also through better matching of heat source temperature in the boiler. Another, study of the solar powered absorption refrigeration system with advance thermal energy storage technology indicates that the Variable mass energy transformation and storage technology can handle low energy storage density and hence, meet the energy storage requirement for solar cooling, heating and dehumidifying demands in Heating, Ventilation and Air Conditioning (HVAC). This technology also provides a better option to use more solar energy for the refrigeration purposes because system is directly driven by solar energy and hence, is more efficient. It also states that under a typical operating condition, the COP of the new system can reach 0.7525 (cooling air) or 0.7555(cooling water), the required solar collector is 66 m^2 (cooling air) or 62 m^2 (cooling water) and the energy storage density of the system is 368.5 MJ/m^3. The comparative study of different types of solar cooling systems in subtropical city reveals the performance of different types of solar cooling systems on the basis of the primary energy consumption which is of the order—*Solar electric compression refrigeration > Solar absorption refrigeration > Solar adsorption refrigeration > Solar solid desiccant cooling > Solar mechanical compression refrigeration.*

The solar electric compression refrigeration and solar absorption refrigeration (with evacuated tubes or flat plate collectors) were the two types of solar cooling systems that could have attractive energy saving potential when compared to conventional electric-driven air-cooled and water cooled vapour compression chiller plants. The year-round energy savings would be from 15.6%–48.3% compared to the conventional electric-driven air cooled refrigeration, while 8.0–43.7% to the water cooled refrigeration.

The introduction of an ejector in the cooling systems can improve the performance of the system. Results showed that that the ejector cooling system designed has a very high COP and can reach 0.5 for a generating temperature of 90 °C, condensing temperature of 28 °C and evaporating temperature of 8 °C. The exergy analysis of the ejector refrigeration system revealed that the largest loss (32%) occurs in the ejector followed by the generator (22%) and the condenser (21%). In addition, the performance of the ejector increases strongly with increased evaporator temperatures and the optimum generating temperature for the selected evaporation temperature of 10 °C is about 80 °C. Another study reveals that that there is an increase in chiller efficiency with an increase in generator temperature but also results in a decrease in solar collector efficiency showing that there exists an optimum generator temperature for maximum collector efficiency.

The comparative tabular representation of the different types of the cooling techniques with methodology adopted and the results obtained is given in the Table 4.

4. Conclusion

The present review study deals with the investigation of the application of the solar energy for different types of the cooling systems in order to reduce the stress on the conventional energy as well to reduce the emission of harmful gases generated due to the burning of fossil fuels for electricity production which is the main source of energy to run the conventional cooling systems. The major conclusions extracted from the review are as.

(a). Many programs of the government and industrial sectors around the world are concentrating on solar energy for different applications to prevent the increase in greenhouse gas emissions into the environment as it is the most promising energy reserve. Fortunately, many studies and reports also revealed the large potential of exploiting solar radiation as a sustainable energy supply for future use.

(b). The detailed study has been carried out to find out the effect of solar energy on the performance of different cooling cycles and the obtained results from the review are summarized below as-

- (i). The comparative study of different types of solar cooling systems in Sub-Tropical city reveals the performance of different solar cooling systems on the basis of the primary energy consumption which is of the order as mentioned below—*Solar electric compression refrigeration > Solar absorption refrigeration > Solar adsorption refrigeration > Solar solid desiccant cooling > - Solar mechanical compression refrigeration*

  Again, it is worthy noted that the solar electric compression refrigeration and solar absorption refrigeration (with evacuated tubes or flat plate collectors) were the two types of solar cooling systems that could have attractive energy saving potential when compared to conventional vapour compression chiller plants. The year-round energy savings would be from 15.6% to 48.3% compared to the conventional electric-driven air cooled refrigeration, while 8.0% to 43.7% to the water cooled refrigeration [2].
- (ii). The results obtained from solar energy powered cascaded refrigeration system revealed that there must be a discrete hot water temperature value needed for the generation of ammonia and below which no ammonia generation takes place. The results obtained also indicate that the exergetic efficiency of the compression system is large when operating temperature differential is of the order of 30 °C in comparison to temperature differential of 45 °C [98]. A theoretical study of a solar driven double effect parallel flow absorption system for sub-zero evaporation temperatures also revealed that COP values are quite acceptable (0.5–0.95) considering the low temperature application, but for better performance the system should be augmented with a hot temperature buffer facility and/or with additional heater [61]. In addition, the simulation study of a solar thermal driven dual-mode, four-bed silica gel–water adsorption chiller revealed that optimum values of COP achieved from two-stage, four-bed and single-stage, four-bed mode are around 0.2 and 0.45, respectively. It is again revealed that cooling capacity increases as the cooling water temperature is lowered [75].
- (iii). The simulation of the solar powered absorption cooling machine with solid absorbent revealed that Coefficient of performance obtained is quite less but with certain modifications in the system there can be an improvement in the performance [18]. The temperature inside the reservoir increases during the desorption process and is slightly higher than ambient temperature because of the fact that condenser temperature is above the ambient temperature. Also, the study to find optimum hot water temperature for absorption solar cooling asserts that there is an increase in chiller efficiency with an increase in generator temperature but also results in a decrease in solar collector efficiency showing that there exists an optimum generator temperature for maximum collector efficiency [24].
- (iv). The solar ejector cooling system revealed that solar ejector cooling has an optimum COP around 0.12 for a

generation temperature of 102 °C and the evaporator temperature of −6 °C for solar radiation at 700 W/m^2 [64]. However, exergy analysis of the ejector refrigeration system revealed that the largest loss (32%) occurs in the ejector followed by the generator (22%) and the condenser (21%). Although, the performance of the ejector increases strongly at increased evaporator temperatures. The optimum generating temperature for the selected evaporation temperature of 10 °C is about 80 °C [70].

(v). A study of solar energy based absorption refrigeration cycle augmented with the compressor to generate the constant energy input revealed that COP is an important parameter for the performance evaluation which is found to be quite higher when compared to conventional system because of higher rate of energy utilization in the new system which eventually leads to reduction in irreversibility of the system. The condenser heat load is also important because it clearly identifies the operational cost of the system [22].

(vi). The first and second law analysis to find out the optimized conditions for refrigeration cycle using solar energy revealed that irreversibility in the cycles is mainly due to the absorber (highest loss 44%), recovery heat exchanger (24%) and rectifier (16%) and such losses could be reduced by using different refrigerant mixture and also through better matching of heat source temperature [31].

(vii). The application of thermal compressors for a single effect and double effect absorption systems driven by solar thermal collectors revealed that that the air-cooled compressors generate more entropy in comparison to water-cooled compressors. The double-effect compressors in both air-cooled and water-cooled cases has higher exergetic efficiency than single effect systems [99].

(viii). A model based on exergy analysis to study solar collector and gas driven water heating and refrigeration plants suggested that maximum second law efficiency is found to be sharp, stressing its practical contribution in the design to make such systems commercially more competitive. The thermal collector fluid capacity rate is also found to be of the highest importance as it is an important finding to assess system scalability [35].

(ix). An investigation of the solar powered absorption refrigeration system with advanced energy storage technology revealed that the COP of the system can reach 0.7525 (cooling air) or 0.7555(cooling water), the required solar collector is 66 m^2 (cooling air) or 62 m^2 (cooling water). The air-cooled condenser elevates collector efficiency and hence requires more solar area where as water cooled condenser results in high efficiency but increases the system's complexity [48].

(x). The study based on Numerical modelling of a solar thermal cooling system revealed that power consumption of a 4.5 kW cooling capacity compression system with the coefficient of performance of 2.6 consumed 15.3 kW h of energy in 9 h of operation whereas solar assisted adsorption cooling system only consumed 10.4 kW h of energy thus cutting down the electric energy demand by 47% [87].

(c). The temperature of the working fluid flowing through the collectors should attain the threshold value to the raise the temperature of the working fluid in the cooling system which virtually is not possible if the temperature falls below the threshold temperature degrading the performance of the system. These conditions are realizable with the use of good quality collectors with selective coatings for better absorption and reflection of the radiations from the surface.

In summary, solar energy operated cooling systems are found to be feasible for industrial or domestic applications, considering that solar radiation is unlimited and available in most parts of the world. Solar cooling technology has positive economic, environmental, and social effects on human life. Solar cooling can also mitigate the existing conventional cooling technologies. Also, solar cooling can reduce the green house effect and CO_2 production and facilitates the establishment of suitable policies in different countries.

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